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Air Cooling in Steam Plant Condenser Using Refrigeration System for Improving Vacuum Pump Performance

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Abstract

Modifying the steam plant condenser by incorporating a vapor compression refrigeration system, which is utilized to chill the water used for cooling the condenser air cooler, is proposed in this work. Hence a considerable drop in air temperature is obtained, and more steam associated with air is condensed resulting in reducing both steam lost and vacuum pump power. A thermodynamic analysis is developed for working out the performance of the proposed hybrid system. The results obtained using this analysis showed that for a condensate subcooling of 3°C the total work saving due to this modification has maximums of 38.5, 33.9 and 28.9% and occur at temperature reduction below the condensate temperature of 12, 10, and 8°C when the temperature of steam admitted to the condenser is 20, 30, and 40°C, respectively. The savings in steam lost through the vacuum pump in these cases amount to 87.7, 84, and 79.2%, respectively. It is recommended to select the decrease in condenser air temperature in the range of 4°C higher than those decreases, at which the minimum total work comes about, as the saving in steam lost becomes fairly greater (5–7%), while the maximum increase in minimum total work is 3%.

Keywords: steam plant condenser, vapor compression refrigeration system, condenser air cooler, vacuum pump, power saving

1. Introduction

The primary purpose of the steam condenser in a steam plant is to convert the turbine exhaust steam into water for reuse in the boiler and to create and maintain vacuum at the turbine exhaust. This is achieved by cold water circulation through tube bundles and condensing the steam on their outer surfaces. Referable to the large reduction in the specific volume on changing from steam to condensate, a vacuum is formed within the condenser. Getting to this vacuum continuously as minimum as possible is very important for the steam plant to attain the highest possible power from the turbine. A rise in the condenser pressure can be produced due to the existence of air, ammonia, and other noncondensable gases [1]. This rise in pressure in the condenser results in lowering the useful turbine power and in turn the plant efficiency degenerates [2–5].

The presence of air lowers the partial pressure of steam and therefore brings down the saturation temperature of steam, which leads to increasing evaporation enthalpy (latent heat), and therefore, more cooling water will be required in the condenser. In addition, this gives rise to increasing condensate subcooling, which is undesirable because it means that the excess heat removed for this purpose serves no useful process and entails providing additional boiler firing [6]. Thus, continuous removal of the air and noncondensable gases from the condenser is very critical for maintaining the plant efficiency possibly high. Beside this task, the air extraction system prevents air blanketing of condenser tubing that could dramatically reduce the heat transfer and stop the condensing process. Also, it reduces the condensate-dissolved oxygen levels that could lead to corrosion of boiler tubing.

The air and noncondensables are extracted from the bottom of the condenser, where the temperature is the lowest, by using vacuum pump. The air and noncondensables sucked by the vacuum pump contain a relatively big amount of noncondensed steam; each kilogram of air can contain more than 2 kg of water vapor [7]. This results in increasing the parasitic work of the pump. In condenser design, it is considered reducing the subcooling of the condensate possibly to the lowest degree to minimize heat removal. For reducing the amount of steam mixed with the air sucked by the air pump, the steam condenser has a portion of tubes near the air pump suction screened off and the condenser tubes in this portion contain the coldest water, and the number of cooling tubes is increased. Hence, the temperature of the air drops below that of the outlet condensate, and more steam associated with the air is condensed so that the mass flow rate of the air and steam mixture drawn by the air pump is reduced and, in turn the power of the dry pump is lowered.

The air extraction system (vacuum pump) can be centrifugal compressor, steam jet ejector air ejector, or water ring pump [8, 9]. A hybrid system of these components is preferred as they offer several advantages toward minimizing venting equipment size and save parasitic consumption [7]. The power required for operating these equipment sorts is dependent on both the rates of air and the noncondensables and noncondensed steam sucked by the vacuum pump. The temperature decrease of the air in the condenser air cooler is limited when the normal cooling water is used. For obtaining considerable drop in air temperature it is proposed in this work to modify the condenser by incorporating a refrigeration system into the condenser. This system is utilized to chill the cooling water, which is used for cooling the air in the condenser rather than the normal cooling water. Of course the use of the refrigerant directly to cool the air exploiting the air cooler as an evaporator for the refrigeration system and circulating the cold liquid refrigerant of this system through the tubes of the air cooler will be more efficient than using the chilled water. However, this configuration calls for redesigning the air cooler to be adapted for receiving and evaporating the refrigerant, which will be expensive for already existing power plants. The latest configuration can be taken into consideration for the design of the new plants.

The concept of using dry ammonia for cooling steam plant condenser is described in Ref. [10]. This concept states that the condenser cooling water is replaced by liquid ammonia, which evaporates as it acquires heat from the condensing steam. The heat of condensation absorbed by the ammonia is rejected in an air cooled condenser of a refrigeration machine into the surrounding atmosphere. This concept was tested and well documented [11–17], with the participation of several major equipment vendors (Baltimore Air Coil, the Trane Company, Curtiss-Wright, CB&I, and Union Carbide). In addition, the concept of using vapor compression refrigeration system (VCRS) combined with steam plant condenser was

studied in detail in [14–19]. The results of these studies showed significant rise in net power as well as efficiency of the steam plant. However, for realizing this combined system, a relatively huge VCRS is required, which leads to material increase of the combined system cost. In the current study, a relatively small VCRS is required as it serves to cool only the air contained in the air cooler of steam plant condenser. This leads to lessening the temperature of the air and water vapor mixture in the air cooler, resulting in condensing some of the water vapor. Hence, the mass flow rate of the air and water vapor mixture extracted from the steam condenser is decreased and consequently the power consumed by the vacuum pump lessens. Meanwhile, water vapor leaving the steam condenser and lost to the surroundings is reduced. In general, this contributes to contracting both the power consumed in the auxiliary systems of the steam plant and the steam consumption.

2. Conception of the modified steam plant condenser

The steam plant condenser is typically a shell and tubes heat exchanger. **Figure 1** depicts a section through a steam plant condenser. The steam leaving the plant turbine enters the condenser at its top with temperature T_{ci} and flows around the outer surface of the cooling tubes in which cooling water is circulated. Heat is transferred from the steam into the cooling water where steam condenses and the condensate is extracted out of the condenser at the bottom with temperature T_{ce} . A segment of the condenser (called air cooler) is shielded for cooling the air associated with the condensate and leaked from the environment. This segment is possibly located near the inlet of cooling water, and the number of the cooling tubes is increased in it, so that the air associated with some water vapor collected in this section is cooled to a temperature T_{vpi} , which is lower than that of the condensate. This will lead to decreasing the mass rate of water vapor, the air pulled by the vacuum pump connected to this segment. As a result, the energy used up by the vacuum pump for extracting the air and its accompanying water vapor to the environment are decreased. In the current study, it is suggested to circulate chilled

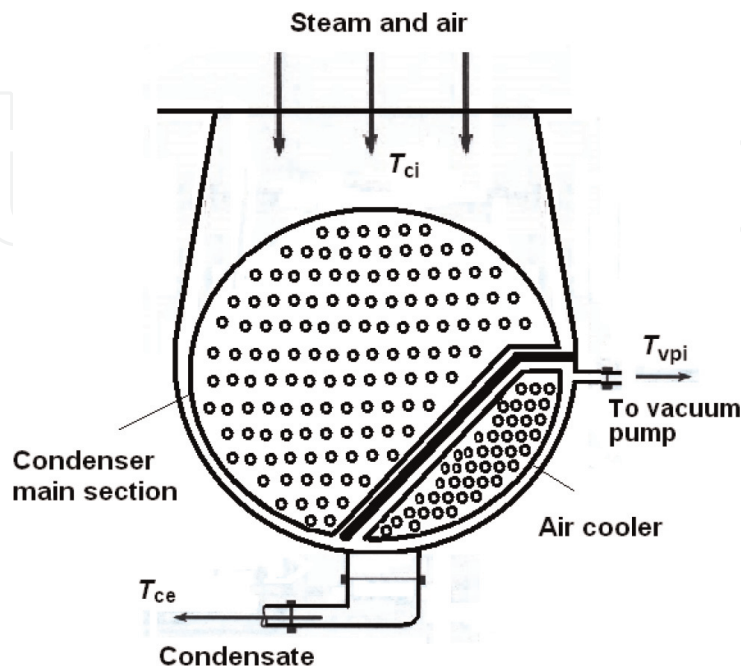


Figure 1.
A schematic section through steam plant condenser.

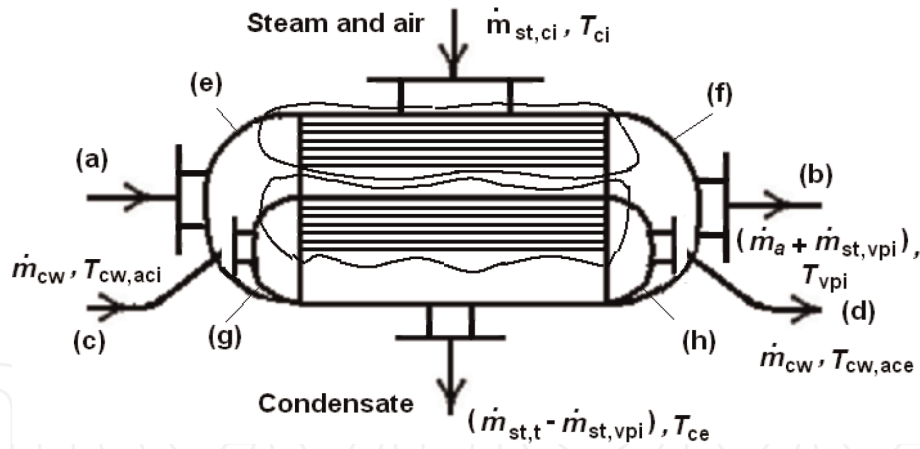


Figure 2.

A configuration of the modified steam plant condenser. a, normal cooling water in; b, normal cooling water out; c, chilled water in; d, chilled water out; e-h, headers.

water within the cooling pipes of the cooling air section, so that the temperature of the air and accompanying steam is further reduced.

Figure 2 illustrates a configuration for the modified steam plant condenser that enables the use of chilled water to lower the temperature of the air cooler section. The cooling water (a) coming from a water body (river, sea, ...) or a cooling tower is fed into the header (e) where it is distributed to the most of the cooling water tubes of the condenser. It helps to cool the steam exhausting out of the plant turbine. This cooling water is called here normal cooling water. While being heated up, it exits the cooling tubes into the header (f). The warmer normal cooling water (b) is extracted from the header (f) and pumped back to the water body or the cooling tower. The chilled water (c) is fed to the header (g) and it flows through the cooling tubes of the air cooling part of the steam condenser. This chilled water acts as a coolant for air along with some water vapor collected in the air cooler. This leads to cooling the mixture of the air and water vapor to a temperature lower than that of the condensate. The temperature of the chilled water is raised as it flows through the cooling tubes due to heat transfer from the mixture of air and steam in the air cooler. As a result, some of the steam contained in the mixture is condensed and flows down the condenser to the hot well. The warm chilled water goes into the header (h), from which it is transmitted backward to a chiller. Finally, the rest of the air and water vapor mixture in the air cooler is sucked by a vacuum pump and expelled into the atmosphere. The vacuum pump is not shown in **Figure 2**.

3. Description of the studied combined system

In the current study, a vapor compression refrigeration system (VCRS) is used to chill the cooling water used for cooling the air cooler of the above described steam condenser. Hence, **Figure 3** shows a schematic of the studied combined system. The refrigeration cycle is made up of a two-stage compression system (c) and (d) (these are also designated by I and II, respectively); a refrigerant condenser (e); two throttling valves (f) and (h); a flash tank (g) for intercooling and flash gas removal; a liquid-line/suction-line heat exchanger (LLSL-HE) (b); and an evaporator (a), which acts at the same time as a heat exchanger for chilling the cooling water used in the air cooler. In this system, the low-pressure refrigerant leaving the evaporator (a) is heated in the liquid suction heat exchanger (b) as it

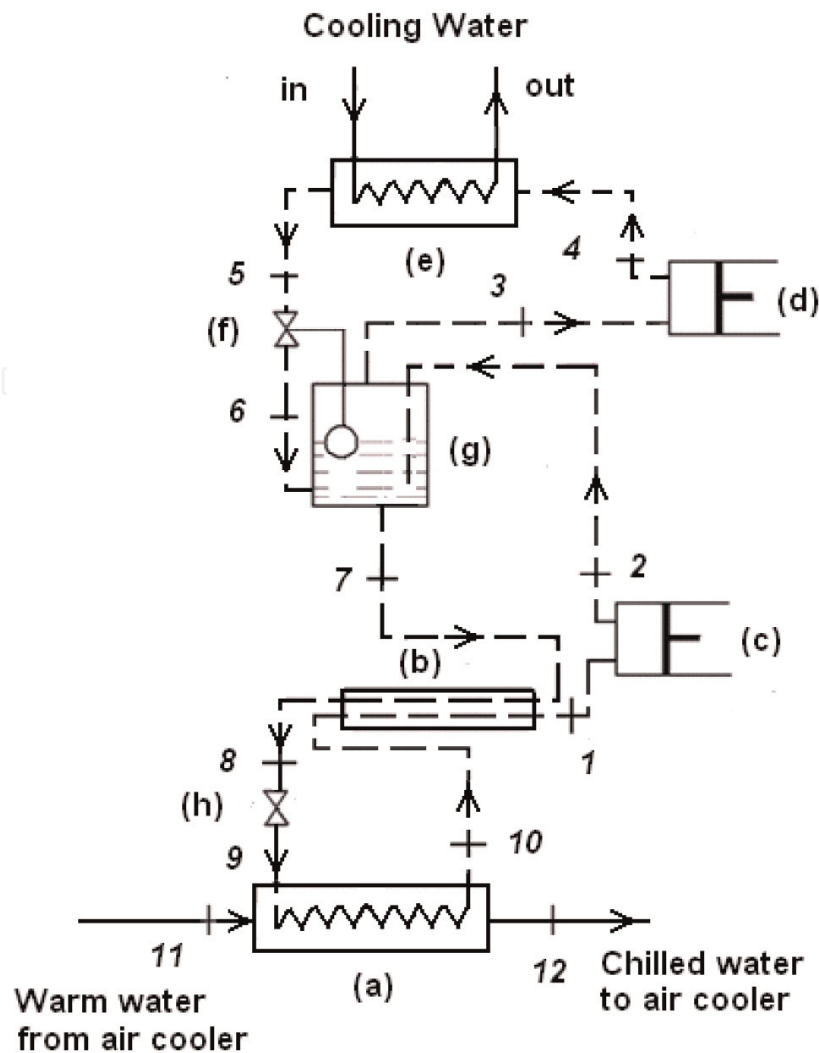


Figure 3.
A schematic of the combined proposed steam plant condenser and vapor compression refrigeration system.
____, water; ---, refrigerant; a, refrigerant evaporator/HE; b, liquid suction HE; c, refrigerant compressor I; d, refrigerant compressor II; e, refrigerant condenser; f, throttle valve I; g, flash chamber; h, throttle valve II.

absorbs heat from the higher-temperature liquid refrigerant coming from the flash chamber (g). It is drawn by the first-stage compressor (c) where its pressure is raised to the intermediate pressure of the flash chamber. It is sent to the flash chamber (g) in which it is mixed with the refrigerant coming out of the throttle valve (f) and it gets cooled. The refrigerant vapor is separated from the liquid refrigerant and it is drawn by the second-stage compressor (d) and its pressure is elevated to the condenser pressure. The high-pressure refrigerant leaving the compressor (d) flows through the refrigerant condenser (e) where it is condensed and it streams further to the throttle valve (f) where its pressure is reduced to the intermediate pressure of the flash chamber, and it is fed into the flash chamber. The liquid refrigerant leaving the flash chamber is fed to the liquid suction HE (b). Here it is cooled as it gives a portion of its sensible heat to the refrigerant vapor departing the evaporator (a). The cooled liquid refrigerant is reduced in pressure to the evaporator pressure on running through the throttle valve (h). The refrigerant is then run to the evaporator/water chiller to cool down the warm water coming from the air cooler. The low-pressure vapor refrigerant leaving the evaporator/water chiller (a), completes the refrigeration cycle. This refrigeration cycle has been selected among the available refrigeration cycles due to its simplicity and relatively high efficiency.

4. Thermodynamic analysis of the studied combined system

For the following analysis of steam and air flow through the plant condenser, it is assumed that both fluids are ideal gases. Since the steam and air flow inside the condenser are at relatively very low pressure, they can be considered behaving to acceptable accuracy as ideal gases. Referring to **Figures 1** and **2**, the steam exhausting the plant turbine (it is usually slightly wet) enters the condenser with temperature T_{ci} and dryness fraction x_{ci} . Knowing the temperature T_{ci} , both the saturation pressure $p_{s,ci}$ (it equals the partial pressure of the steam) and specific volume $v_{s,ci}$ of the saturated steam at condenser inlet can be fixed. Given the steam mass flow rate $\dot{m}_{st,t}$ entering the turbine, the mass flow rate $\dot{m}_{st,ci}$ of the steam at condenser entry can be calculated by:

$$\dot{m}_{st,ci} = x_{ci} \dot{m}_{st,t} \quad (1)$$

The volume flow rate $\dot{V}_{st,ci}$ at inlet to the condenser, by Dalton's law, is equal to the volume flow rate $\dot{V}_{a,ci}$ of the associated air [20] and is obtained from the following equation:

$$\dot{V}_{st,ci} = \dot{V}_{a,ci} = \dot{m}_{st,ci} v_{s,ci} \quad (2)$$

The partial pressure $p_{a,ci}$ of the air at entry to the condenser is calculated by:

$$p_{a,ci} = \frac{\dot{m}_a R_a (T_{ci} + 273.15)}{\dot{V}_{a,ci}} \quad (3)$$

Inserting Eqs. (1) and (2) into Eq. (3) and designating the mass ratio $\dot{m}_a/\dot{m}_{st,t}$ by β , it follows that:

$$p_{a,ci} = \frac{\beta R_a (T_{ci} + 273.15)}{x_{ci} v_{s,ci}} \quad (4)$$

The total pressure $p_{c,t}$ at condenser entry is equal to the sum of the partial pressure $p_{a,ci}$ of the air and the saturation pressure $p_{s,ci}$ of the steam entering the condenser. It is taken constant throughout the condenser, since the velocity of steam flow is small. Hence, the total absolute pressure $p_{c,t}$ inside the condenser is given as:

$$p_{c,t} = p_{s,ci} + p_{a,ci} \quad (5)$$

The condensate temperature T_{ce} at condenser outlet is usually ΔT_{ce} lower than the steam temperature T_{ci} at condenser inlet (i.e., $T_{ce} = T_{ci} - \Delta T_{ce}$). Knowing the temperature T_{ce} , the saturation pressure $p_{s,ce}$ and specific volume $v_{s,ce}$ of the steam corresponding to this temperature can be determined. If the condenser is not screened, then the partial pressure $p_{a,ce}$ of the air leaving the steam condenser with condensate is given as:

$$p_{a,ce} = p_{c,t} - p_{s,ce} \quad (6)$$

The volume flow rate $\dot{V}_{a,ce}$ of the air to be dealt by the vacuum pump at the exit of the steam condenser can be determined by:

$$\dot{V}_{a,ce} = \frac{\dot{m}_a R_a (T_{ce} + 273.15)}{p_{a,ce}} \quad (7)$$

Hence, the mass flow rate $\dot{m}_{st, ce}$ of steam associated with the air is assessed by:

$$\dot{m}_{st, ce} = \frac{\dot{V}_{a, ce}}{v_{s, ce}} \quad (8)$$

Dividing both sides of Eq. (8) by $\dot{m}_{st, t}$ and inserting Eq. (7) into Eq. (8), it follows that:

$$\gamma = \frac{\dot{m}_{st, ce}}{\dot{m}_{st, t}} = \frac{\beta R_a (T_{ce} + 273.15)}{p_{a, ce} v_{s, ce}} \quad (9)$$

In the case of screening the steam condenser and cooling the air, the temperature T_{vpi} of the air and water vapor mixture sucked by the vacuum pump is ΔT_{vpi} lower than the condensate temperature T_{ce} at steam condenser outlet (i.e., $T_{vpi} = T_{ce} - \Delta T_{vpi}$). Knowing the temperature T_{vpi} , the corresponding saturation pressure $p_{s, vpi}$ and specific volume $v_{s, vpi}$ can be fixed. The partial pressure $p_{a, vpi}$ and volume flow rate $\dot{V}_{a, vpi}$ of the air and mass ratio δ ($\dot{m}_{st, vpi} / \dot{m}_{st, t}$) at inlet of the vacuum pump can be determined using Eqs. (6), (7) and (9), respectively, by replacing the subscripts ce by vpi and γ by δ . Accordingly, it follows that $p_{a, vpi}$, $\dot{V}_{a, vpi}$, and δ are given by:

$$p_{a, vpi} = p_{c, t} - p_{s, vpi} \quad (10)$$

$$\dot{V}_{a, vpi} = \frac{\dot{m}_a R_a (T_{vpi} + 273.15)}{p_{a, vpi}} \quad (11)$$

and

$$\delta = \frac{\dot{m}_{st, vp}}{\dot{m}_{st, t}} = \frac{\beta R_a (T_{vpi} + 273.15)}{p_{a, vpi} v_{s, vpi}} \quad (12)$$

Regarding the VCRS, the p - h diagram of its cycle is illustrated in **Figure 4**. The numerals of **Figure 4** correspond to the points 1–10 given in **Figure 3**. It is to be considered here that the refrigerant condenser is cooled exactly as it is conducted with the steam plant condenser. Consequently, it is assumed here that the refrigerant leaving the refrigerant condenser has a temperature equal to that of the condensate in the steam plant condenser (i.e., $T_5 = T_{ce}$). Considering this fact and

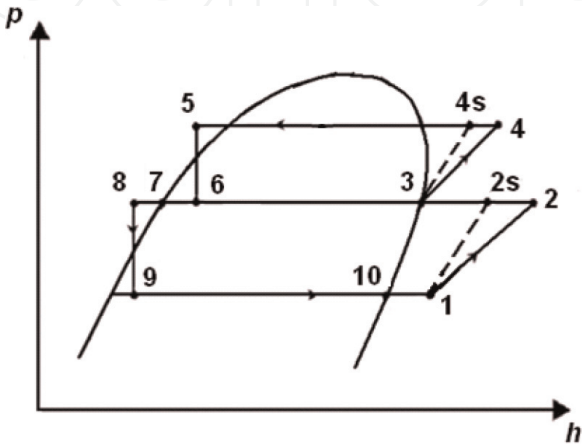


Figure 4.
 p - h diagram of the VCRS cycle of the studied refrigeration system.

knowing the subcooling $\Delta T_{rc,sub}$ of the refrigerant condenser, the pressure of the refrigerant in the condenser (e) can be fixed. The evaporator temperature $T_9 = T_{10}$ is defined according to the temperature required by the chilled water at inlet to the air cooler of the steam condenser. T_9 as well as T_{10} is ΔT_e less than the temperature $T_{cw,aci}$ (i.e., $T_9 = T_{10} = T_{cw,aci} - \Delta T_e$) of the chilled water entering the steam condenser air cooler. Hence, the pressure of the refrigerant in the evaporator is the saturated one corresponding to the temperature T_9/T_{10} . Knowing the refrigerant pressures in the evaporator (a) and condenser (e), the adiabatic efficiencies $\eta_{rco,I}$ and $\eta_{rco,II}$ and mechanical efficiencies $\eta_{m,rco,I}$ and $\eta_{m,rco,II}$ of the compressor I (c) and II (d), respectively, and the effectiveness ε_{LLSL} of the LLSL-HE, the states of the different points of the VCRS cycle can be determined as explained in any refrigeration text book (e.g., [21]).

For obtaining the value of the mass ratio ζ ($\dot{m}_{cw}/\dot{m}_{st,t}$) of the chilled water flow rate through the air cooling segment and steam flow rate crossing the turbine, **Figure 5** shows the mass flow rates of air mixture and chilled water, and their temperatures through this segment. An energy balance for the air cooler leads to the following equation:

$$\begin{aligned} \dot{m}_a c_{p,a} (T_{ce} - T_{vpi}) + (\dot{m}_{st,aci} - \dot{m}_{st,vpi}) LH \left[\text{at saturation pressure } p_{s,vpi} \right] \\ = \dot{m}_{cw} C_w (T_{cw,ace} - T_{cw,aci}) \end{aligned} \quad (13)$$

It is to be noticed here that the sensible heat of the condensed steam in the air cooler was neglected as it is relatively very small, and $\dot{m}_{st,aci}$ is equal to $\dot{m}_{st,ce}$.

Solving Eq. (13) to get \dot{m}_{cw}/\dot{m}_a and multiplying both sides of the resulted equation by $\dot{m}_a/\dot{m}_{st,t}$, the ratio ζ ($\dot{m}_{cw}/\dot{m}_{st,t}$) is obtained as:

$$\zeta = \frac{\dot{m}_{cw}}{\dot{m}_{st,t}} = \frac{\beta c_{p,a} (T_{ce} - T_{vpi}) + (\gamma - \delta) x LH \left[\text{at saturation pressure } p_{s,vpi} \right]}{c_w (T_{cw,ace} - T_{cw,aci})} \quad (14)$$

For determining the mass ratio $\dot{m}_{r,I}/\dot{m}_{cw}$ of the refrigerant and cooling water, a heat balance is performed for the evaporator/water HE (a), which yields to:

$$\dot{m}_{r,I} (h_{10} - h_9) = \dot{m}_{cw} C_w (T_{cw,ace} - T_{cw,aci}) \quad (15)$$

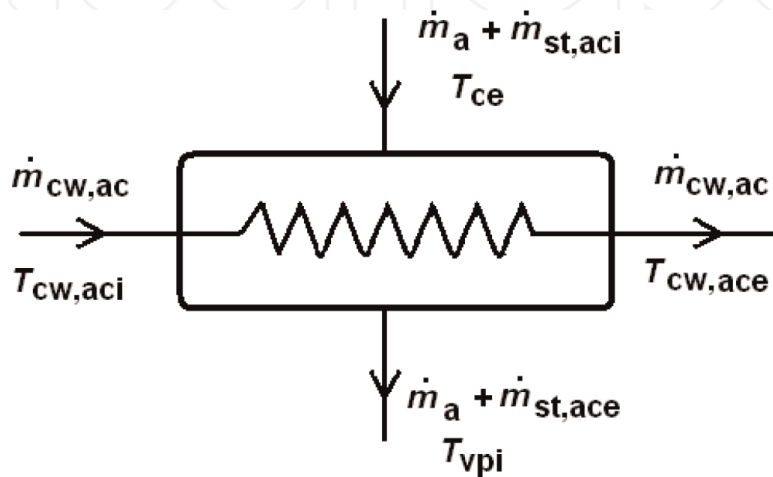


Figure 5.
Mass flow and temperature of air, steam, and chilled water through the air-cooling segment.

From which it follows that:

$$\frac{\dot{m}_{r,I}}{\dot{m}_{cw}} = \frac{C_w (T_{cw,ace} - T_{cw,aci})}{(h_{10} - h_9)} \quad (16)$$

Multiplying both sides of Eq. (16) by the mass ratio $\dot{m}_{cw}/\dot{m}_{st,t}$, the mass ratio ξ_I ($\dot{m}_{r,I}/\dot{m}_{st,t}$) is worked out as:

$$\xi_I = \frac{\dot{m}_{r,I}}{\dot{m}_{st,t}} = \zeta \frac{C_w (T_{cw,ace} - T_{cw,aci})}{(h_{10} - h_9)} \quad (17)$$

The mass ratio ζ is obtained from Eq. (14). A heat balance of the flash chamber (Figure 3) yields:

$$\dot{m}_{r,II} h_6 + \dot{m}_{r,I} h_2 = \dot{m}_{r,I} h_7 + \dot{m}_{r,II} h_3 \quad (18)$$

It follows from Eq. (18) that:

$$\frac{\dot{m}_{r,II}}{\dot{m}_{r,I}} = \frac{(h_2 - h_7)}{(h_3 - h_6)} \quad (19)$$

Multiplying both sides of Eq. (19) by $\dot{m}_{r,II}/\dot{m}_{r,I}$, it follows that the mass ratio ξ_{II} ($\dot{m}_{r,II}/\dot{m}_{st,t}$) of the refrigerant flow rate through the compressor II to the steam flow rate of the steam turbine is given by:

$$\xi_{II} = \frac{\dot{m}_{r,II}}{\dot{m}_{st,t}} = \frac{\dot{m}_{r,II}}{\dot{m}_{r,I}} \times \frac{\dot{m}_{r,I}}{\dot{m}_{st,t}} = \xi_I \frac{\dot{m}_{r,II}}{\dot{m}_{r,I}} = \xi_I \frac{(h_2 - h_7)}{(h_3 - h_6)} \quad (20)$$

From Eqs. (17) and (20), the mass ratio ξ_t can be found as:

$$\xi_t = \frac{\dot{m}_{r,t}}{\dot{m}_{st,t}} = \xi_I \xi_{II} = \zeta \frac{C_w (t_{cw,ace} - t_{cw,aci})}{(h_{10} - h_9)} \left[1 + \frac{(h_2 - h_7)}{(h_3 - h_6)} \right] \quad (21)$$

$$\xi_t = \frac{\dot{m}_{r,t}}{\dot{m}_{st,t}} \quad (22)$$

The characteristic parameters describing the performance of the refrigeration cycle are given as:

$$q_{rc} = \xi_{II} (h_4 - h_5) \quad (23)$$

$$w_{rco,I} = \frac{\xi_I}{\eta_{m,co,I}} (h_2 - h_1) = \frac{\xi_I}{\eta_{m,rco,I} \eta_{i,rco,I}} (h_{2s} - h_1) \quad (24)$$

$$w_{rco,II} = \frac{\xi_{II}}{\eta_{m,rco,II}} (h_4 - h_3) = \frac{\xi_{II}}{\eta_{m,rco,I} \eta_{i,rco,II}} (h_{4s} - h_3) \quad (25)$$

$$w_{rco,t} = (w_{rco,I} + w_{rco,II}) \quad (26)$$

$$q_e = \xi_{II} (h_{10} - h_9) \quad (27)$$

$$COP = \xi_{II} (h_{10} - h_9) / [\xi_I (h_{2s} - h_1) / \eta_{co,I} \eta_{m,co,I} + \xi_{II} (h_{4s} - h_3) / \eta_{co,I} \eta_{m,co,II}] \quad (28)$$

The hybrid system proposed in the current work can lead to two benefits, the first benefit is a decrease in the mass flow rate of steam lost in venting the air from the SPC, and the second one is a reduction in mass flow rate of air and steam mixture drawn by the vacuum pump and hence the pump power is lowered.

However, a new energy consumption comes out, which is the total work ($w_{\text{rco,t}}$) of the two compressors employed in the VCRS. To be capable to judge the goodness of the proposed hybrid system, both the energies used for venting the air from SPC and for operating the compressors of the refrigeration system should be determinable. The total work $w_{\text{rco,t}}$ of the two refrigerant compressors can be determined using Eqs. (24)–(26). As for the work consumed for venting process, there are some vacuum pumps that can be utilized for this function, among which centrifugal compressor is the most effective and efficient instrument for performing this task. It is selected here only for the sake of judgment of the hybrid system goodness. In venting the air and steam mixture out of the steam condenser without air cooling, the specific vacuum pump (compressor) work $w_{\text{vp,wac}}$ referred to each kilogram of steam flow through the steam turbine can be expressed by aid of any thermodynamics text book (e.g., [20]) as:

$$w_{\text{vp,wac}} = \frac{k_m R_m T_{ce} (\beta + \gamma)}{(k_m - 1) \eta_{m, \text{vp}} \eta_{i, \text{vp}}} \left[\left(\frac{p_{\text{atm}}}{p_t} \right)^{\left(\frac{k_m - 1}{k_m} \right)} - 1 \right] \quad (29)$$

where k_m and R_m are the isentropic exponent and gas constant, respectively, of the air steam mixture. They are given by [20]:

$$k_m = \frac{\beta C_{p,a} + \gamma C_{p,st}}{\beta C_{v,a} + \gamma C_{v,st}} \quad (30)$$

$$R_m = \frac{\beta R_a + \gamma R_{st}}{\beta + \gamma} \quad (31)$$

When the steam condenser is fitted with air cooler, the air extracting compressor work w_{vp} is given by:

$$w_{\text{vp,ac}} = \frac{k_m R_m T_{vpi} (\beta + \delta)}{(k_m - 1) \eta_{m, \text{vp}} \eta_{i, \text{vp}}} \left[\left(\frac{p_{\text{atm}}}{p_t} \right)^{\left(\frac{k_m - 1}{k_m} \right)} - 1 \right] \quad (32)$$

k_m and R_m are calculated in this case by aid of Eqs. (30) and (31), respectively, by replacing γ by δ .

The total specific work w_t ($w_{\text{rco,t}} + w_{\text{vp,ac}}$) employed for cooling and driving out steam condenser air is calculated by summing up Eqs. (26) and (32).

5. Results and discussion

The thermodynamic analysis developed in Section 4 for predicting the condensation rate in the steam plant condenser due to air cooling is first validated with the experimental data of reference [22]. In this work, experiments were conducted in a 2-m-long square cross-sectional channel ($0.34 \text{ m} \times 0.34 \text{ m}$) to study the heat and mass transfer in the condensation of water vapor from humid air. The air flowing inside the channel was cooled by cold water flowing outside and adjoining only one side ($0.34 \text{ m} \times 2 \text{ m}$) of the channel. Experimental data were obtained from five tests at various operating conditions as shown in **Table 1**. The thermodynamic analysis in Section 4 was slightly modified to be adapted for using the data given in **Table 1** for computing the condensation rate. For solving the equations in this analysis and finding out the rate of condensation, the commercial computer package EES [23] was used. The thermal properties of the humid air at different conditions were found using the built-in functions available in the EES package. The condensation

Test	Inlet humid air temperature $T_{ha,in}$ (°C)	Velocity of humid air v_{ha} (m/s)	Inlet relative humidity of humid air ϕ	Average cooling flux $q_{c,av}$ (kW/m ²)
1	82.66	1.46	100	7.3
2	80.61	2.02	100	9.0
3	79.13	2.52	97.83	10
4	78.73	3.01	87.35	11.1
5	75.02	3.59	96.55	12.5

Table 1.
Experimental conditions from Ref. [22].

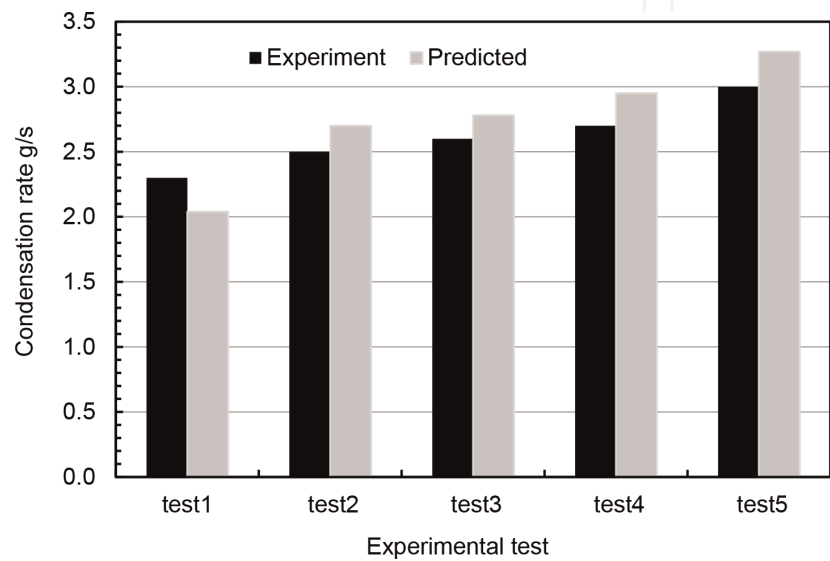


Figure 6.
Comparison of the predicted and experimental values of condensation rate for all the tests of Ref. [22].

rate computed from the present model is displayed in **Figure 6**, which shows satisfactory agreement with the experimental data of Ref. [22] since the maximal discrepancy does not exceed 10%.

The thermodynamic analysis developed in Section 4 for performance prediction of the combined system proposed in this work necessitates knowing some basic design and operational data, which is listed in **Table 2**. It is to be noticed here that the values of isentropic and mechanical efficiencies of the compressors involved in this study have been selected close to the practical values of compressors in use in industry [24]. The results presented hereafter are based on these data. All parameter values given in **Table 2** will be kept unchanged except for the case where the effect of a specific parameter is to be examined; it is handled as a variable.

The refrigerant of the refrigeration system is selected to be ammonia. The physical properties of air, water, steam, and ammonia needed for computation are predicted using the built-in functions of the commercial computing package EES [23], which is used for solving the equations of the analysis of Section 4.

In **Figure 7**, the mass ratio γ is plotted versus the temperature T_{ci} for values of ΔT_{ce} of 1, 3, and 5°C. It is seen from **Figure 7** that γ runs linearly with very low rate with T_{ci} . This can be explained as follows: since β is constant, the mass rate of steam condensed because cooling the air depends mainly on ΔT_{ce} and it is very little dependent on T_{ci} . Of course the condensed steam rate in the air cooler is a pit higher at higher T_{ci} . For constant β and T_{ci} , the amount of steam associated with the

Parameter	Value
Mass ratio (β)	0.0003
Compressor isentropic efficiency ($\eta_{i,rc0,I}, \eta_{i,rc0,II}, \eta_{i,vp}$)	0.85
Compressor mechanical efficiency ($\eta_{m,rc0,I}, \eta_{m,rc0,II}, \eta_{m,vp}$)	0.75
Dryness fraction of steam entering the SPC (x_{ci})	0.9
Effectiveness of the liquid suction HE (ϵ_{LLSL})	0.8
Temperature difference ($T_{ci} - T_5$), °C	0
Temperature difference ($T_{cw,ace} - T_{cw,aci}$), °C	4
Temperature difference ($T_{cw,aci} - T_{10}$), °C	4
Temperature difference ($T_{cw,ace} - T_{vpi}$), °C	4
Temperature difference ΔT_{ces} , °C	3
Subcooling of the refrigerant condenser ($\Delta T_{rc,sub}$), °C	3

Table 2.
Basic design and operational data of the proposed combined steam plant condenser and refrigeration system.

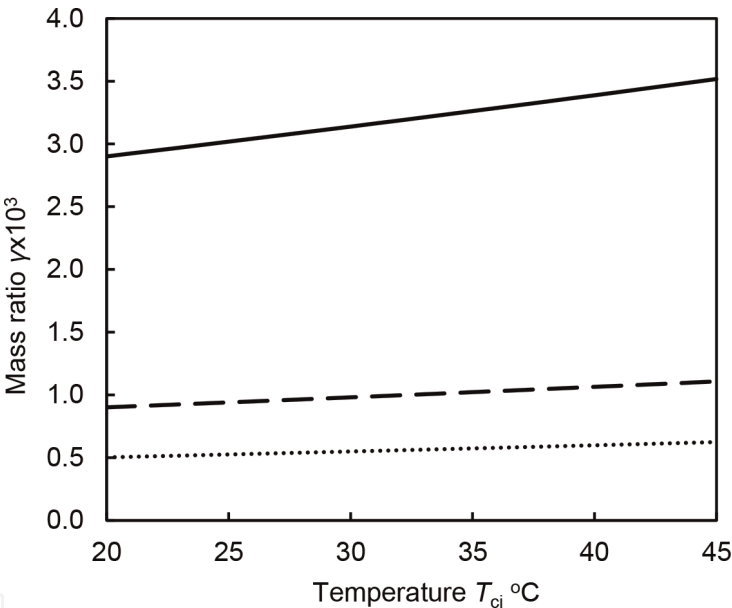


Figure 7.
Effect of condensate temperature on the extracted steam mass rate by the vacuum pump without air cooler.
——, $\Delta T_{ce} = 1^\circ\text{C}$; — — —, $\Delta T_{ce} = 3^\circ\text{C}$; , $\Delta T_{ce} = 5^\circ\text{C}$.

condenser air and, in turn, the rate of condensed steam decrease progressively with ΔT_{ce} . This accounts for the remarkable drop in γ with an increase in ΔT_{ce} as shown in **Figure 7**. In contrast, the amount of steam associated with the condenser air and consecutively the rate of condensed steam for constant β and ΔT_{ce} are almost unvarying with T_{ci} . This explains the very low rate of increase in γ with rising T_{ci} .

It is worth noting here that in **Figures 8–14**, which will be displayed in this section, the small values of temperature difference ΔT_{vpi} close to zero represent the case in which there is virtually no air cooler is employed. These values are not practical as the refrigeration system will be useless. Yet these values are included in these images just for illumination. In **Figure 8**, the mass ratio δ is drawn against the temperature difference ΔT_{vpi} for temperature T_{ci} of 20, 30, and 40°C. It is seen from **Figure 8** that δ declines with an increase in ΔT_{vpi} where the rate of declination is relatively high at small values of ΔT_{vpi} and it decreases progressively with ΔT_{vpi}

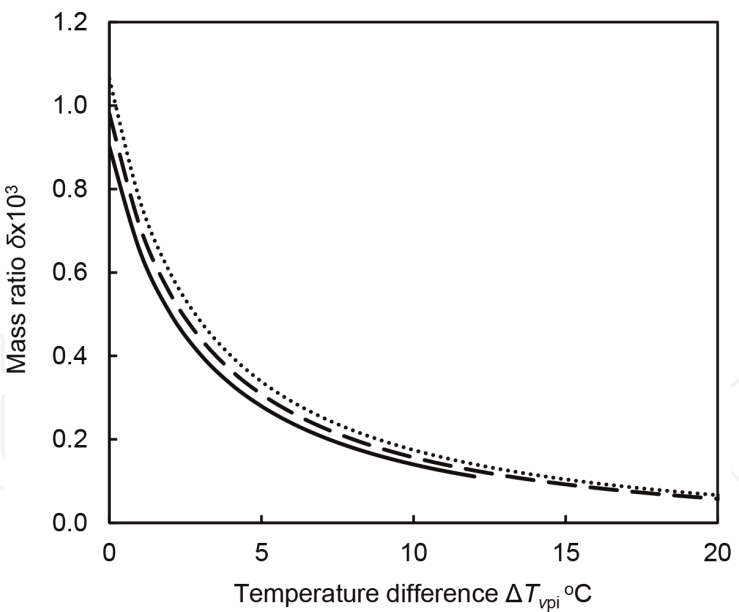


Figure 8.
 Dependence of the mass of steam associated with air on the air temperature at vacuum pump entrance. —, $T_{ci} = 20^{\circ}\text{C}$; — —, $T_{ci} = 30^{\circ}\text{C}$;, $T_{ci} = 40^{\circ}\text{C}$.

and becomes immaterially small for values of ΔT_{vpi} greater than 12°C . This is brought about due to the large drop in steam content in the air and likewise the rate of steam condensed as the temperature T_{vpi} falls down. It is also seen from **Figure 8** that T_{ci} has almost negligible effect on δ as the amount of steam mixed with the condenser air and in turn the rate of condensed steam for constant β and ΔT_{vpi} are almost invariable with T_{ci} .

Figure 9 illustrates the saving percentage $(\gamma - \delta) \times 100/\gamma$ in the steam amounts to be condensed in the air cooler and not sucked by the vacuum pump as steam. It is clear from **Figure 9** that this saving has a reversed trend to that of the mass ratio δ , it is equal to zero at $\Delta T_{vpi} = 0$, and it increases steeply with ΔT_{vpi} . The rate of increase in this saving with ΔT_{vpi} falls increasingly with the rise in ΔT_{vpi} where it becomes inconsiderably small at ΔT_{vpi} of 12°C .

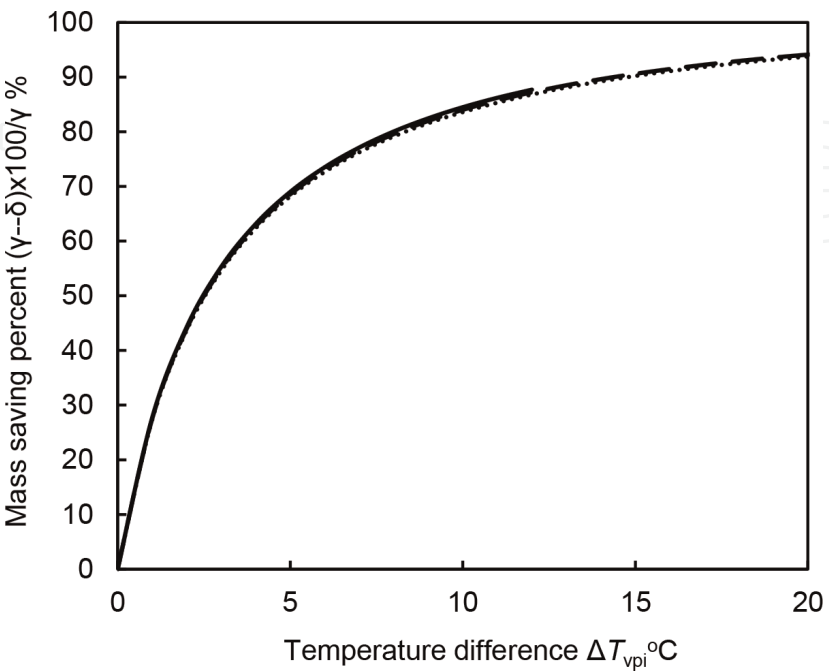


Figure 9.
 Saving percentage in steam mass associated with air when using air cooler. —, $T_{ci} = 20^{\circ}\text{C}$; — —, $T_{ci} = 30^{\circ}\text{C}$;, $T_{ci} = 40^{\circ}\text{C}$.

It follows from **Figures 8 and 9** that the amount of steam to be condensed in the air cooler is relatively high at small values of ΔT_{vpi} and the rate of increase in this amount falls progressively with ΔT_{vpi} . Therefore, the amount of the cooling chilled water and in turn the refrigerant needed for chilling the cooling water takes the same trend of the percentage saving $(\gamma - \delta) \times 100/\gamma$ (see **Figure 9**). **Figures 10 and 11** show the mass ratios ζ and ξ_t , respectively, as a function of ΔT_{vpi} .

In **Figure 12**, the coefficient of performance COP of the refrigeration system is plotted versus the temperature difference ΔT_{vpi} for T_{ci} of 20, 30, and 40°C. **Figure 12** discloses distinctly that COP decreases sharply with ΔT_{vpi} . This is ascribed mainly to the falling value of the evaporator temperature of the

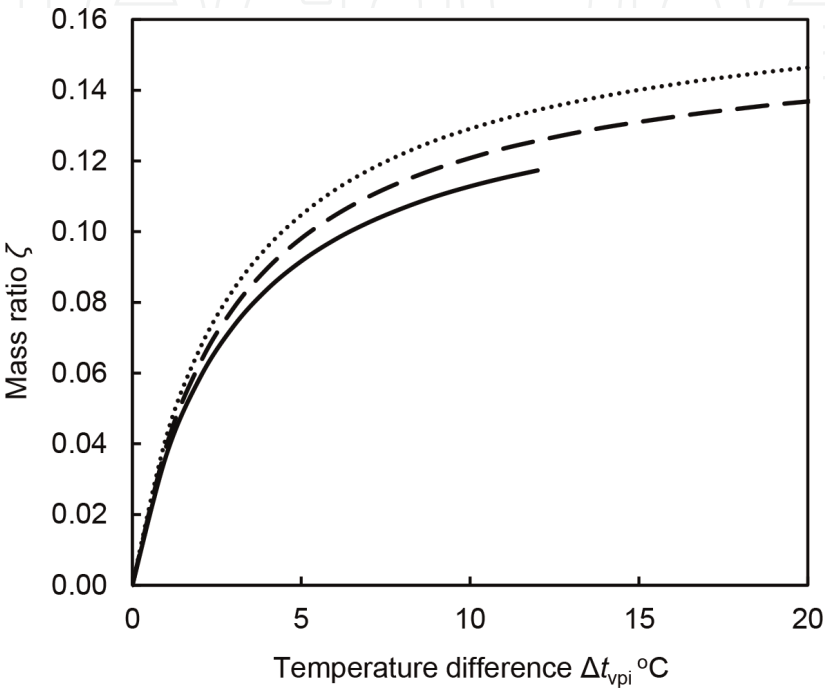


Figure 10.
Relationship between the mass of chilled cooling water required for steam condenser air cooler and the air temperature at vacuum pump entrance. —, $T_{ci} = 20^\circ\text{C}$; — —, $T_{ci} = 30^\circ\text{C}$;, $T_{ci} = 40^\circ\text{C}$.

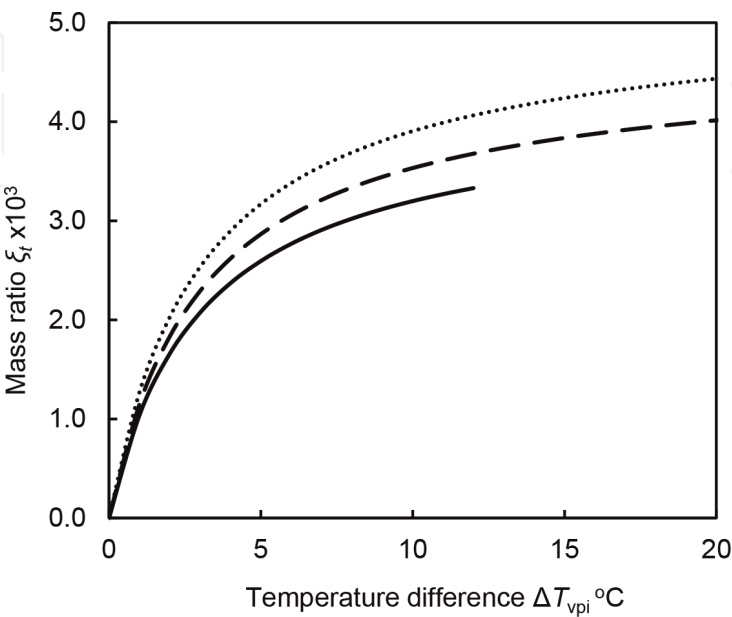


Figure 11.
Effect of the air temperature at vacuum pump entrance on refrigerant mass rate when using air cooler. —, $T_{ci} = 20^\circ\text{C}$; — —, $T_{ci} = 30^\circ\text{C}$;, $T_{ci} = 40^\circ\text{C}$.

refrigeration system and the increasing temperature difference between the refrigerant condenser and the evaporator. The temperature T_{ci} has almost a negligible effect on COP as the temperature difference between refrigerant condenser and evaporator alters with ΔT_{vpi} and not with T_{ci} .

The specific works $w_{vp,wac}$, $w_{rco,t}$, $w_{vp,ac}$ as well as w_t are plotted in the diagrams of **Figure 13** versus ΔT_{vpi} for temperature T_{ci} of 20, 30, and 40°C. Although $w_{vp,wac}$ is independent of ΔT_{vpi} , it is represented on the diagrams of **Figure 13** as horizontal lines for the sake of comparison. It can be seen from **Figure 13** that $w_{vp,wac}$ decreases with an increase in T_{ci} , which is caused mainly due to the drop in pressure ratio of the vacuum pump. The specific work $w_{vp,ac}$ is reduced steeply with ΔT_{vpi} until a value of ΔT_{vpi} around 12°C; then, the rate of decrease in $w_{vp,ac}$ diminishes remarkably. This can be interpreted as follows: on the one hand, at low temperature difference ΔT_{vpi} , the amount of steam mixed with air is relatively high as can be seen from **Figures 8 and 9**, which results in relatively high mass rate of the mixture of air and water vapor flowing through the vacuum pump, and therefore, greater pump work $w_{vp,ac}$ is obtained. As ΔT_{vpi} is raised, the amount of steam flowing with air dwindles and so the mass flow rate through the pump declines, which gives rise to decreasing the pump work $w_{vp,ac}$. On the other hand, the pressure ratio through the vacuum pump is raised with ΔT_{vpi} and hence the work $w_{vp,ac}$ is increased. However, the increase in $w_{vp,ac}$ is relatively small at low values of ΔT_{vpi} compared to the work decrease due to the drop in pump mass flow rate. Therefore, the pump work $w_{vp,ac}$ falls off with relatively high rate, as ΔT_{vpi} grows. As the mass flow induced by the pump declines and the pressure ratio through the pump grows with increasing ΔT_{vpi} , the effect of the former parameter diminishes, while the effect of the latter parameter grows up and the net result is a considerable drop in the rate of decrease in $w_{vp,ac}$. The specific work $w_{rco,t}$ rises almost linearly with ΔT_{vpi} . This is attributed mainly to the declination of the refrigeration system COP (see **Figure 12**). The rate of increase in $w_{rco,t}$ is almost independent of the temperature T_{ci} .

The saving percentage $(w_{vp,wac} - w_t) \times 100/w_{vp,wac}$ in work by using refrigeration system for cooling the air contained in the steam condenser is plotted in **Figure 14** versus ΔT_{vpi} for T_{ci} of 20, 30, and 40°C. It is seen from **Figures 13 and 14**

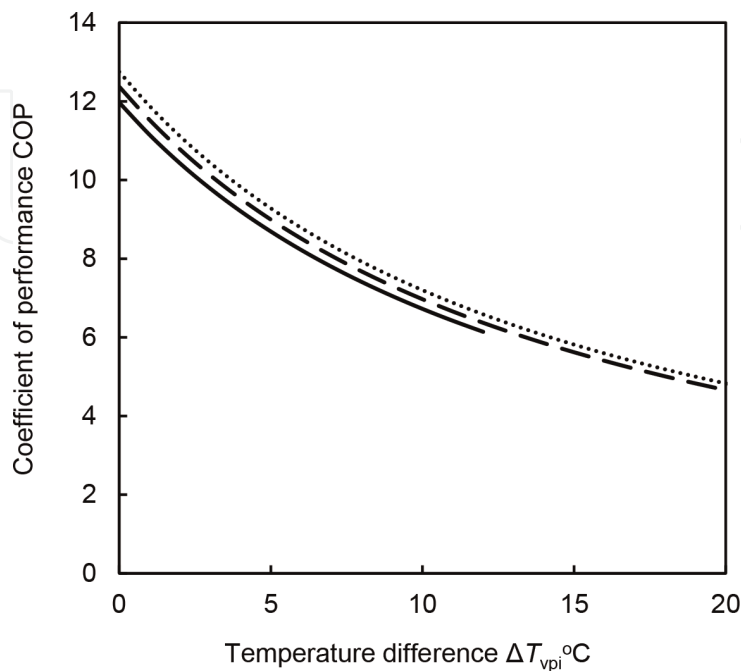


Figure 12.
Effect of the air temperature at vacuum pump entrance on coefficient of performance of the refrigeration system.
——, $T_{ci} = 20^\circ\text{C}$; — — —, $T_{ci} = 30^\circ\text{C}$; , $T_{ci} = 40^\circ\text{C}$.

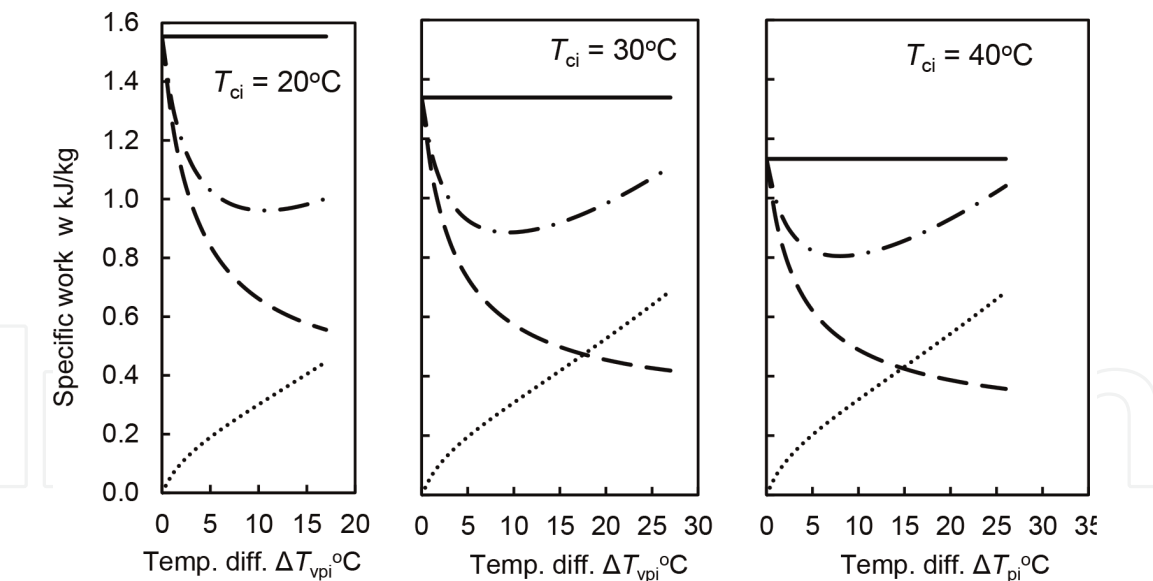


Figure 13. Specific work dependence on the air temperature at vacuum pump entrance. —, $w_{vp,wac}$; — — —, $w_{vp,ac}$; , $w_{rco,t}$; — · —, w_t .

that the sum of the specific works $w_{vp,ac}$ and $w_{rco,t}$ (i.e., the total specific work w_t) has a minimal value (maximum saving in the total work sum w_t). This minimum value depends on the value of T_{ci} and it is less than the corresponding $w_{vp,wac}$ according to the temperature T_{ci} by 38.5, 33.9, and 28.9% at ΔT_{vpi} of 12, 10, and 8°C for T_{ci} of 20, 30, and 40°C, respectively. Also, it is seen from **Figures 13** and **14** that w_t is maximally higher than the minimal value corresponding to T_{ci} by 3% when ΔT_{vpi} is 4°C higher than its value at which the minimal total specific work occurs. On the contrary, the saving in the steam lost increases depending on T_{ci} in the range of 5–7%. Therefore, it is more advantageous to choose values for ΔT_{vpi} higher than those at which the minimal total work occurs in the range of 4°C, as it results in fairly less lost steam rate to be drawn by the vacuum pump and inconsiderable increase in the total work. Higher values than 4°C cause inconsiderably small increase in saving the lost steam, but the total work is significantly raised.

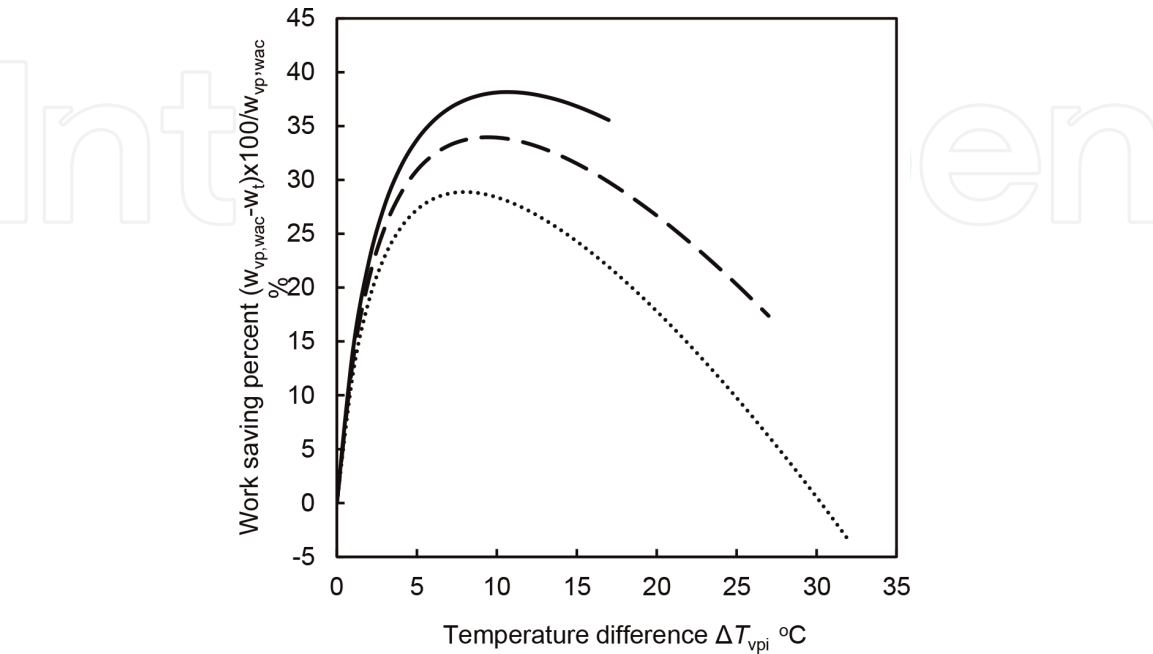


Figure 14. Saving percent in specific work due to refrigeration cooling of steam condenser air cooler. —, $T_{ci} = 20^\circ\text{C}$; — — —, $T_{ci} = 30^\circ\text{C}$; $T_{ci} = 40^\circ\text{C}$.

It is to be mentioned here that the parameters γ , δ , ζ , ξ_t , $w_{vp,wac}$, $w_{vp,ac}$, $w_{rco,t}$, and w_t are displayed in **Figures 7, 8, 10, 11, and 13** for the value for β of 0.0003. These parameters are directly proportional to the mass ratio β . This is explained as follows: the mass of steam mixed with each kilogram of air and condensed and in turn the amount of cooling normal/chilled water used for cooling a kilogram of air and condensing the steam as well as the amount of refrigerant utilized for chilling the cooling water are dependent only on the initial and final air temperatures of the cooling process. Therefore, the parameters mentioned above are directly proportional to the mass ratio β . On the contrary to that, the saving percentages $(\gamma - \delta) \times 100/\gamma$ and $(w_t - w_e) \times 100/w_e$ in the amount of steam to be condensed and total work, respectively, are independent of β .

6. Conclusions

The current work is concerned with the use of vapor compression refrigeration system (VCRS) for chilling cooling water used with the air cooler of the steam plant condenser (SPC). A thermodynamic analysis is developed for working out the performance of the hybrid system of VCRS and SPC. The results obtained using this analysis showed that subcooling of the SPC condensate can cause considerable reduction in steam rate associated with the air induced by the vacuum pump. However, this is possibly avoided as it represents heat loss in the condensate heat content, which should be compensated in the plant boiler. In addition, the results of this work led to drawing the following conclusions for condensate subcooling of 3°C, which represents a reasonable and practical subcooling of the condenser condensate.

1. Temperature reductions of the condenser air of 5, 10, and 15°C below the condensate temperature result in reducing steam rate lost in venting air from the condenser relative to the loss when using no air cooler, by around 69, 85, and 90%, respectively.
2. The total work saving when using chilled water for cooling the air in the condenser air cooler from that in case of no air cooling is applied, has maximums of 38.5, 33.9, and 28.9% and occurs at temperature decrease below the condensate temperature of 12, 10, and 8°C when the temperature of steam admitted to the condenser is 20, 30, and 40°C, respectively. In these cases, the savings in steam lost in venting process amount to 87.7, 84, and 79.2%, respectively.
3. Selecting the reduction in condenser air temperature in the range of 4°C higher than that temperature reduction, at which the minimum total work occurs, is very advantageous where the saving in steam lost becomes fairly greater while the saving in total work is slightly lower than the minimum total works; in the range of 5–7 and maximally 3%, respectively.

Nomenclature

COP	coefficient of performance of the refrigeration system
C_p	specific heat capacity at constant pressure (kJ/kg K)
C_w	specific heat capacity of water (kJ/kg K)
h	specific enthalpy (kJ/kg)
k	isentropic exponent

LH	specific latent heat (kJ/kg)
\dot{m}	mass flow rate (kg/s)
p	pressure (kPa)
q	relative heat transfer rate (kW/kg)
R	gas constant (kJ/kg K)
T	temperature (°C)
v	specific volume (m ³ /kg)
V	air velocity (m/s)
\dot{V}	volume flow rate (m ³ /s)
x	dryness fraction (–)
w	specific work (kJ/kg)

Greek letters

β	mass ratio $\dot{m}_a/\dot{m}_{st,t}$ (–)
γ	mass ratio $\dot{m}_{st,ce}/\dot{m}_{st,t}$ (–)
δ	mass ratio $\dot{m}_{st,vp}/\dot{m}_{st,t}$ (–)
ΔT_{ce}	temperature difference of steam at condenser inlet and exit (K/°C)
ΔT_e	temperature difference of chilled water at air cooler inlet and saturated refrigerant in VCRS evaporator (K/°C)
$\Delta T_{rc,sub}$	subcooling in the refrigerant condenser
ΔT_{vpi}	temperature difference of condensate in hot well and air entering the vacuum pump (K/°C)
ε	heat exchanger effectiveness (–)
ζ	mass ratio $\dot{m}_{cw}/\dot{m}_{st,t}$ (–)
η_i	isentropic efficiency of the compressor (–)
η_m	mechanical efficiency of the compressor (–)
$\xi_b \ \xi_{lb} \ \xi_t$	mass ratios $\dot{m}_{r,I}/\dot{m}_{st,t}$, $\dot{m}_{r,II}/\dot{m}_{st,t}$, and $(\dot{m}_{r,I} + \dot{m}_{r,II})/\dot{m}_{st,t}$, respectively (–)
φ	relative humidity (–)

Subscripts

a	air
ac	with air cooler
ace	air cooler exit for chilled water
aci	air cooler inlet for chilled water
atm	atmospheric
av.	average
c	steam condenser, cooling
ce	steam condenser outlet
ci	steam condenser inlet
cw	chilled water
e	evaporator
ha	humid air
i	isentropic
in	inlet
m	air and steam mixture, mechanical
r	refrigerant
rc	refrigerant condenser
rco	refrigerant compressor
s	saturated steam
st	steam

<i>t</i>	steam turbine, total pressure, total work
<i>vp</i>	vacuum pump
<i>vpi</i>	vacuum pump inlet
<i>w</i>	water
<i>wac</i>	without air cooler
1–10	state numbers of the refrigerant of the refrigeration cycle
I, II	refrigerant compressor no. I and II, respectively

Abbreviations

LLSL-HE	liquid-line/ suction-line heat exchanger
SPC	steam plant condenser
VCRS	vapor compression refrigeration system

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